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EFFECT OF WHITE METAL COATING ON BORE BEARING PROFILE PERFORMANCE OF (ROTATING JOURNAL INSIDE SHELL)

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ABSTRACT

This paper was focusing on the study the effect of white metal alloys coating on rotary journals because these journals are subjected to corrosion. And usually, the corroded bearings are losing the operational properties, for this reason, these bearings needed for rehabilitation again by coating the rotary parts with the appropriate thickness. In order to ensure effective conductivity via the two solid zones the rotary journals and coating layers (base metal and white metal alloys coating) to get rid of the oil temperature. White metal coating with different depth was used between mating surface and the heat conduction equation was applied at both regions of the rotating part with suitable boundary condition. The results show that the optimum non-dimensional thickness for white metal coating is (0.25), which improve the properties of the metals by giving the best get rid of oil temperature. Also, it increases the non-dimensional friction force by increasing the non-dimensional thermal conductivity. As well as the increasing of Non-dimensional maximum oil and bush temperature which in turn decreases the Sommerfeld number, while increasing the non-dimensional white metal coating thickness lead to increase the Sommerfeld number and non-dimensional friction coefficient.

KEYWORDS: Corrosion, Rotary Journals, Sommerfeld number, Theoretical Study & White Metal Coating

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Original Article

1. INTRODUCTION

The using of metals return to 4,500 BC due to appear the needing for strong, easy fabricates tools and equipment. The metals applications have become widespread in many fields such as machine, construction field, aerospace, medical field and rare metals with the high value used in jewellery and accessories, **(Riskin, 2008).**

And Journal Bearing is one of metal application that used as a machine part to support, and radially location a rotating shaft. Bearing's action and effectiveness affect the successful process of the systems/mechanisms. Thus, bearing materials should be chosen sensibly, to succeed these systems run and meet the action anticipations. Rolling contact bearings have lower friction than sliding contact bearings. But the use of sliding contact bearings is inevitable. They have their specific benefits and are used highly. Journal bearings are used in manufacturing apparatuses, engines, and automobile industry, hydraulic turbines, electric generators, steam and gas turbines, compressors and other machines used in power, oil, gas, and petrochemical industries. Journal bearings are also called plain bearings, sleeve bearings, and fluid film bearings. Selection of material for bearing applications depends on the type of bearing, type of lubrication and environmental conditions **(Hamrock et al. 2004).**

Journal bearing material should have a group of properties such as compatibility, conformability, embed-

ability, fatigue strength, cavitation erosion resistance, and corrosion resistance. There is no specific material has all the requirements to be a good bearing material. Thus, a compromise and mix of these properties are required for successful the action under a specific operation condition **(Sturk and Whitney 2013).** Rubbing of the shaft and bearing material against each other should not produce localized welds which leads to scoring or seizure or scuffing. This inherent tendency is called compatibility. When there is a slight misalignment in the bearing assembly, the bearing material should undergo a little deformation to minimize the stressintensities and maintain oil film thickness. This ability is called conformability. Embed-ability is the ability to embed hard particles in the surface of the bearing material and thus reduce any abrasive damage to both shaft and bearing. The ability of any bearing material to resist scoring bases on the above three factors. Compatibility is difficult to quantify, whereas conformability and embed-ability vary inversely with hardness.

In addition to the above list of characteristics, there are some desirable mechanical characteristics of bearing materials; few are the compressive strength, fatigue strength, low coefficient of friction (COF), low coefficient of thermal expansion, high thermal conductivity, good wettability, sufficient hardness, enough elasticity, its availability, and cost. Lubrication of moving parts in journal bearings plays an important role in the wear and frictional behavior, **(Babu, et.al., 2015).**

And the corrosion problem is one of the major issues that all materials or products, plants, constructions, and buildings made of metals are suffering from physical corrosion during the use period. A general overview of different types of corrosion resulting from mechanical, thermal, chemical, electrochemical, microbiological, electric, and radiation, **(Maaß, et.al., 2011).**Corrosion is a reaction (chemical and electrochemical) between a material and environments which lead to producing a deterioration for the material properties. In general, there are many types of corrosion such as general corrosion which resulting from the environmental issues. The second corrosion type is localized corrosion which caused by variation in chemical or physical surroundings between connecting sites. Also, Bacterial corrosion is a type of corrosion that caused by the production of bacteria with a contact metal on the surface of the steel. As well as, Galvanic/Dissimilar Metal is a type of corrosion that caused by contacting dissimilar metals together. However, the corrosion in any type will reduce the metal performance and if the metal used in any machine the corrosion will reduce the machine performance. **(Langill, 2006)**

As mentioned previously, there are many types of corrosion as well as there are many reasons that lead to cause corrosion. On the other hand, many researchers try to reduce or control corrosion. They found many methods for controlling the corrosion such as Barrier Protection (Paint, Powder Coatings, Galvanizing), Cathodic Protection, Corrosion Resistant Materials, **(Langill, 2006).** And the white metal coating is one of these methods that used to protect metal from corrosion. White metals are metals or alloys that have a light color such as lead, tin, zinc, antimony, cadmium, and bismuth. These metal mixed in different proportion in order to achieve a desired goal or need, in general, the white metals alloys contain either tin (Sn) or lead (Pb) as a base. The tin-based alloys containing 7-10% antimony (Sb) and 3-5% copper (cu). And typical lead-based alloys containing 14% antimony (Sb), 10% tin (Sn), 5% copper (cu), and remainder lead (Pb). And for each type of these alloys, there are different advantages, the lead base alloysare cheap but it shows inferior wear and brittleness compared to tin-based alloys so this alloys usually used in high-class machineries such as turbine and marine's gearing with lead regarded as additive material, **(Branagan, 2015).**

Herschman and Basil (1932) they investigated that the wear resistance and the mechanical properties exceptionwear of different white metalbearing alloys at the temperature (20C°-200C°). Resistance to wear was studied at

20 C° only where the tin-base alloys showed higher resistance to wear than lead-base and cadmium-zinc alloys, **(Bose, 2017).**

Moreover, the white metal coating in the rotating part is very essential to improve the performance of such part operating under therom hydrodynamic operation (THD). The good example of such subject is the profile bore bearing which working under laminar flow regime. The profile bore bearing consisting of a shaft, housing, and lubricant. And the bearing material must be selected carefully so as to carry the applied load without seizure, fatigue, distortion, corrosion, and wear and the lubricant should operate satisfactorily with them.

A white metal coating has many advantages for applying such as supporting the load smoothly in the rotary element, decreasing the friction force to the lowest values, keeping the shaft concentric in the case of zero load, maintaining high wear and friction resistances and transmitting more heat to the surrounding area due to good thermal conductivity. And due to its low melting point, it is easily casting, sticking strongly to the metal base and have a reasonable elastic limit which has a good impact on the final characteristics. The thickness of the coating is generally between (2-0.25)mm. Normally wear happened in the rotary surface especially in contact points so as to white metal coating is used for re-healing the mating surface for the new life, **[(Frederick and Capper, 1976) &(Herschman and Basil, 1932)].**

Figure 1: White Metal Coating

Table (1) show different types of white metal alloys that had been used in order to determine the effect of white metal coating on the performance of the rotating part in contact points under suitable hydrodynamic lubrication. This study was carried out forevaluating the temperature distribution and fluid viscosity for both fluid and bush. The theoretical analysis based on decoupled energy and Reynolds equations to obtain the conventional performance parameters using two-dimensional approach of (**McCallion, et.al., 1970**), the heat conduction equation for the solid domain is used with suitable boundary condition for white metal coating.

Table 1: Below Shows Some white Metal Alloys Specifications [(Herschman and Basil, 1932)& (Lyman, 1967)]

No.	Alloy name	Components Proportion %				K w/m c°
		Sn	Pb	Sb	Others	
1	BS 3332	---	94	6	---	28.90
2	BS 3332	---	96	4	---	30.57
3	BS 3332	63	37	---	---	50.25
4	BS 3332	5	80	15	---	24.28
5	ASTM B23	92	---	---	8 Zn	58.62
6	DIN 1708	---	91	9	---	26.80

Lyman (1967) said that during the choosing of white metal material should take in the consideration Continuity of service, Bonding properties, Cooling facilities, Lubrication, Cleanliness and Maintenance schedule for the bearing use. For example, the continuous using for a bearing in a harsh environment without regular maintenance will require different Babbitt and lubrication comparison with a bearing in intermittent use in a clean, light-duty environment.

Kingsbory, (1933) investigated the theoretical and experimentalthermal effect in a Couette flow. This work considered as starting THD analysis. The thermal viscosity variation, momentum, and energy equation were solved. While **Swift (1937)** investigated the effective viscosity to obtained the effective bearing performance. He assumed that the largest heat amount is transferred out of the bearing through lubricant. On the other hand, **Dowson (1962)** Established THD solution by solving energy and Reynolds equation both across and along the film thickness. Dowson and March (1967) Conducted THD solution for journal bearing in two dimensional model across and along the film lubricant in addition to the effect of cavitation.

McCallion et al. (1970) presented simplified method which uncouples the Reynolds and energy equations by neglecting the pressure term in the energy equation, he compared his results with the exact numerical methods and with the experimental result of Dowson and March (1967), the axial heat in the bush was neglected, only radial and circumferential were considered.

All previous study focused on the effect of heat transfer in the lubricant as the important factor which is strongly effected lubricant pressure and temperature which in turn affect the lubricant viscosity, load carrying capacity and machines part cooling parameters. As a result of the eccentric rotating shaft inside the bush two regiondevelopednamed converging and diverging zones, in the diverging part the lubricant mixed with gas bubbles or gas cavities due to partially existing lubricant before the flow reformation take place such case is called cavitation which is communally exists in a journal bearing. The first work appeared in 1886 by Reynolds as a theoretical work to predict the pressure distribution across the fluid film bearing. However, this study focus on the using of the white metal coating which carried out by the centrifugal casting with suitable thickness and machining process to get the depth and roughness required and study its influences mathematically depending on many equations.

Studying the Modell Mathematically

Many Researchers **[(Hussain, et.al., 1992), (Mistry, et.al., 1992), (Boncompain, et.al., 1980), (Crosby, 1980), (Majumdar and Saha, 1974), (Safar and Szeri, 1974) and (McCallion, et.al., 1970)]** used uncoupling technique of energy and Reynolds equation. Where they applied a theoretical approach to analysis the effect white metal coating as a concentration of the energy equation and Reynolds equations with its considerations for presenting analysis and evaluating the results. The theoretical analysis is made by the different type of white metal alloys as given in table (1) with various thicknesses to get the optimum thickness values.

The non-dimensional Reynolds equation was suggested by **McCallion, et.al. (1970)** as following:

$$\frac{\partial}{\partial\theta}\left(\frac{\overline{H}}{\overline{\mu}}\frac{\partial\overline{P}}{\partial\overline{\theta}}\right)+\left(\frac{R}{L}\right)^2\frac{\partial}{\partial\overline{y}}\left(\frac{\overline{H}}{\overline{\mu}}\frac{\partial\overline{P}}{\partial\overline{y}}\right)=6\frac{\partial\overline{H}}{\partial\overline{\theta}} \qquad (1)$$

The non-dimensional parameters are as mentioned by **[(Dowson and March 1967), (McCallion, et.al., 1970), (Majumdar and Saha, 1974) & (Hussain, 1992)]**

$$\overline{H} = \frac{h}{c}, \overline{\theta} = \frac{x}{R}, \overline{y} = \frac{y}{L}, \overline{\mu} = \frac{\mu}{\mu_{in}}, \overline{P} = \frac{P}{\mu_{in}\omega}\left(\frac{c}{R}\right)^2 \quad (1.1)$$

If the lubricants feeding at the center line then the Bearing Configuration **(**B.C) as following:

$$\bar{P}(\overline{\theta}, \overline{y}) = 0 \; at \overline{\theta} = 0, \bar{P}(\overline{\theta}, \overline{y}) = \frac{\partial \overline{P}}{\partial \overline{\theta}} = 0 \; at \overline{\theta} = \theta_{c,} P(\overline{\theta}, \overline{y}) = 0 \; at \overline{y} = 1$$

The thickness film is getting as following:

$$\overline{H} = 1 + \varepsilon cos\theta \quad (2)$$

The load capacity is carrying as following:

$$\mathrm{W_t} = \int_0^{\theta_c}\int_0^1 \overline{\mathrm{P}} \; \sin\theta \, d\overline{y} \, d\overline{\theta} \quad (3)$$

$$\mathrm{W_r} = \int_0^{\theta_c}\int_0^1 \overline{\mathrm{P}} \; \cos d\overline{y} \, d\overline{\theta} \quad (3.1)$$

$$W^* = \sqrt{W_t{}^2 + W_r{}^2} \quad (3.2)$$

From Figure (2) W_t is the component along the center line and W_r is the component normal to the line of center and W^* is the total load carrying capacity.

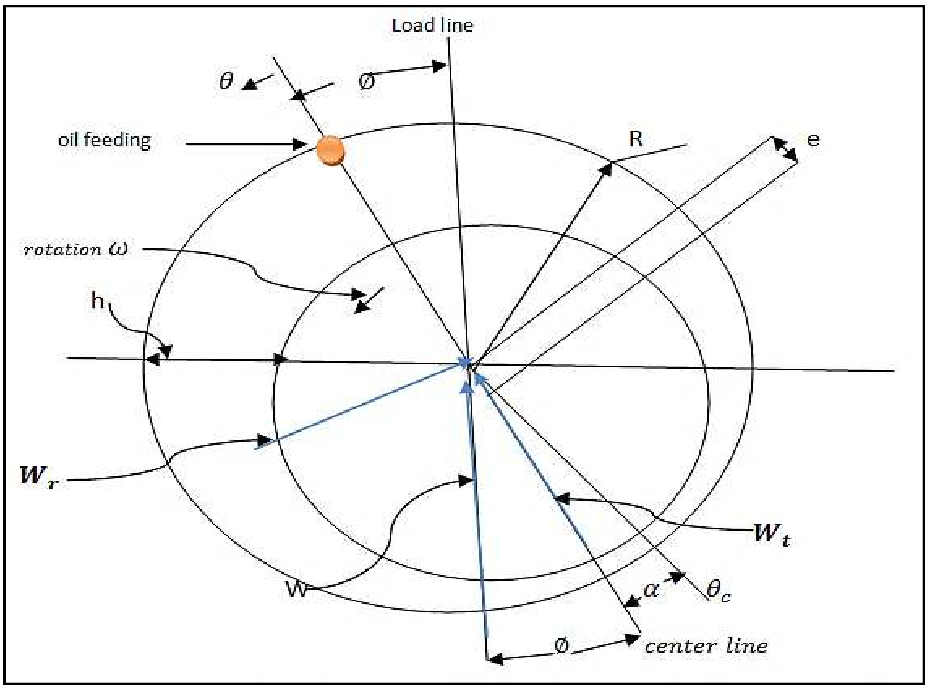

Figure 2: Center Line Oil Feeding Journal Bearing Configuration

The Dimensional Load Capacity

$$W = W^* \mu_{in} UL \left(\frac{R}{c}\right)^2 \quad (4)$$

The Sommerfeld number is determined as following:

$$S = \frac{\mu_{in} NLD \left(\frac{R}{c}\right)^2}{W} \quad (5)$$

$$whereNistherotationalspeedin(^{rev}/_{sec}) = \frac{\omega}{2\pi}$$

The Sommerfeld number investigated by another form,

$$S = \frac{1}{\pi W^*}$$

The angle between the load line and the center line called attitude angle which investigated by eq. (6)

$$\emptyset = tan^{-1}\frac{W_t}{W_r} \tag{6}$$

The non-dimensional friction coefficient is as **(Hussain, 1992)** mentioned:

$$\overline{\eta} = \eta\left(\frac{R}{c}\right) \tag{7}$$

$$\eta = \frac{F_r}{w}$$

Where, η: a friction coefficient.

While, **(Hussain, et.al., 1996)** the non-dimensional side leakage is investigated by eq. (8):

$$\overline{Q} = \frac{2Q}{ULc}, whereQ = \frac{1}{12}\int_0^{x_c}\frac{h}{12}\frac{\partial P}{\partial y}dx \tag{8}$$

The energy equation which suggested by **McCallion, et.al. (1970)** was identifying the temperature distribution within the oil film considering decoupling with Reynolds equation as following:

$$\overline{H}^2\overline{u}\frac{\partial\overline{T}}{\partial\overline{\theta}} = Pe\left(\frac{\partial^2\overline{T}}{\partial\overline{z}^2}\right) + \overline{\mu}\left(\frac{\partial\overline{u}}{\partial\overline{z}}\right)^2 \tag{9}$$

The B.C of equation (9) is investigated depending on **[(McCallion, et.al., 1970), (Suganami, and Szeri, 1979), (Sinhasan, and Chandrawat, 1989) & (Hussain, 1992)]** as following:

On the fluid bush interface, the bush temperature is equal to the oil temperature as following:

$$T|_{z=0} = T_b|_{R=R_i}$$

On the shaft interface, the temperature is same as the mean temperature of the bush inner surface.

On the groove the temperature is equal to T_{in}

Thermal conduction equation will be applied across both the white metal alloy coating and the thickness of the base bush material.

In order to determine the heat conduction across the white metal alloys, the following formula is used:

$$\frac{\partial^2\,\overline{T}_C}{\partial\overline{R}_C{}^2} + \frac{1}{\overline{R}_C}\frac{\partial\overline{T}_C}{\partial\overline{R}_C} + \frac{1}{\overline{R}_C^2}\frac{\partial^2\overline{T}_C}{\partial\overline{\theta}^2} = 0 \tag{10}$$

And (R_c) is determined based on both (R) and (t) as shown in eq. (11)

$$R_c = R + t \tag{11}$$

Where (t) is the non-dimensional Babbitt thickness

The heat conduction of the thickness of the bush base material can be determined by this equation as following:

$$\frac{\partial^2 \bar{T}_b}{\partial \bar{R}_b{}^2} + \frac{1}{\bar{R}_b}\frac{\partial \bar{T}_b}{\partial \bar{R}_b} + \frac{1}{\bar{R}_b^2}\frac{\partial^2 \bar{T}_b}{\partial \bar{\theta}^2} = 0 \qquad (12)$$

According to **[(McCallion, et.al., 1970), (Suganami, and Szeri, 1979), (Sinhasan, and Chandrawat, 1989), (Hussain, 1992) &(Hussain, et.al., 1996)]** there are many boundary conditions for heat conduction equation as following:

The B.C for the fluid-bush interface can be gotten from the continuity of heat flux from the following equations:

$$K_o \left.\frac{\partial T}{\partial Z}\right|_{Z=0} = K_c \left.\frac{\partial T_c}{\partial R_C}\right|_{R=R_{bi}} \qquad (13)$$

For the non –dimensional form the equation becomes:

$$\left.\frac{\partial \bar{T}_c}{\partial \bar{R}_c}\right|_{R=R_{bi}} = \left(\frac{K_o}{K_c}\frac{\mu_i \omega}{C_p \rho \{c/R\}^2}\frac{R}{\bar{H} c T_{in}}\right)\left.\frac{\partial \bar{T}}{\partial \bar{Z}}\right|_{\bar{Z}=0} \qquad (14)$$

On the bush base material and coating interface the B.C is:

$$K_c \left.\frac{\partial \bar{T}_c}{\partial \bar{R}_c}\right|_{R=R_c} = K_b \left.\frac{\partial \bar{T}_b}{\partial \bar{R}_b}\right|_{R=R_c} \qquad (15)$$

On the outer part of the housing (R=R_{bo}) the free convection and radiation hypothesis give the B.C as following:

$$\left.\frac{\partial \bar{T}_b}{\partial \bar{R}}\right|_{R=R_{bo}} = Bib\left(\bar{T}_b\big|_{R=R_{bo}} - \bar{T}_a\right) \qquad (16)$$

Where, $Bib = \frac{h_b R}{K_b}$

The heat generation in the cavitation zone is less than the pressurized zone due to the presence of air bubbles or air cavities. According to **[(Suganami and Szeri,1979), (Sinhasan and Chandrawat 1989), (Mistry, et.al.,1992) & (Hussain, et.al., 1996)]** the fluid mixture and air properties may be represented as the mixture, so the equivalent properties can be as following:

$$K_E = \left(1 - \frac{\bar{H}_m}{\bar{H}}\right)K_a + K_0\frac{\bar{H}_m}{\bar{H}} \qquad (17)$$

$$\bar{\mu}_E = \bar{\mu}_0 \frac{\bar{H}_m}{\bar{H}}$$

[(Suganami and Szeri,1979), (Sinhasan and Chandrawat 1989), (Mistry, et.al.,1992) & (Hussain, et.al., 1996)] depending on the heat flux B.C, the equivalent heat flux of mixture in the cavitation zone can be written as following:

$$\rho_E C_{PE} = \rho_o C_{PO}\frac{\bar{H}_m}{\bar{H}}$$

$$\rho_a << \rho_o$$

The non-dimensional viscosity–temperature relationship is as following **[(Hussain, et.al., 1992) & (Hussain, et.al., 1996)]**:

$$\bar{\mu} = \frac{\mu}{\mu_{in}} = \frac{A\left(\frac{T}{S}\right)^{T/B}}{\mu_{in}}$$

Where:

A, B and S are constants, their values depend upon the oil grade. The viscosity used in the calculation as the mean viscosity in the circumferential direction **[(Crosby, 1980), (Hussai, et.al., 1992) & (Mistry, et.al., 1992)].**

Lubricant Mixing

In the feeding groove the recycled oil Q_r will mix with the inlet oil Q_{in} and flow mixture will be determined from the energy balance [(**Hussai, et.al., 1992**) & (**Hussai, et.al., 1996**)]

$$C_P Q_m T_{mix} = C_P Q_{in} T_{in} + C_P Q_r T_r \tag{18}$$

$$Q_m = Q_r + Q_{in}$$

$$T_{mix} = \frac{Q_{in} T_{in} + Q_r T_r}{Q_r + Q_{in}}$$

$$Q_{in} = ULh_{mix}$$

$$Q_r = Q_{in} - Q$$

Where: Q is the side leakage

Solution Procedure

- Assume initial temperature which is equal to inlet temperature.
- The initial viscosity values equal to the inlet oil viscosity.
- Solve the energy equation for pressure and cavitation zone with suitable B.C.
- Estimate the bush temperature from heat conduction equation with its suitable B.C
- Repeat steps 3 and 4 till convergence of 0.001 occurs.
- Calculate the mean viscosity according to the new fluid temperature distribution.
- Calculate Pressure distribution and the load.
- Calculate the white metal temperature distribution with its suitable B.C.

RESULTS AND DISCUSSION

This study was focused on the effect of white metal coating on profile bore bearing with different alloys, so the results will include the theoretical and mathematical investigation of white metal effect on the rotary part of journal bearings as following.

Figure 3 represents the friction force (Fr (N)) versus the non-dimensional thermal conductivity (K) for different non-dimensional thickness of white metal coating as following (0.15, 0.2, 0.25, 0.4 & 0.5). Figure 3 shows that the friction force increases with increasing the non-dimensional thermal conductivity. As well as the white metal coating allows to high heating transition via the white metal in the radial direction to the surrounding parts. And because of the high heating transition, the lubricant is remaining cool which lead to increase the convective capacity. Figure 3 clearly shows the best thickness is 0.25as it gives minimum friction force which is lead to decrease shear stress to its minimum limits. Also, this thickness gives a good indication for the friction force (Fr (N)) versus the non-dimensional thermal conductivity (K) curve because there is no sudden increasing or decreasing in the curve so the shear stress could be estimated easily comparison

with other coating thicknesses.

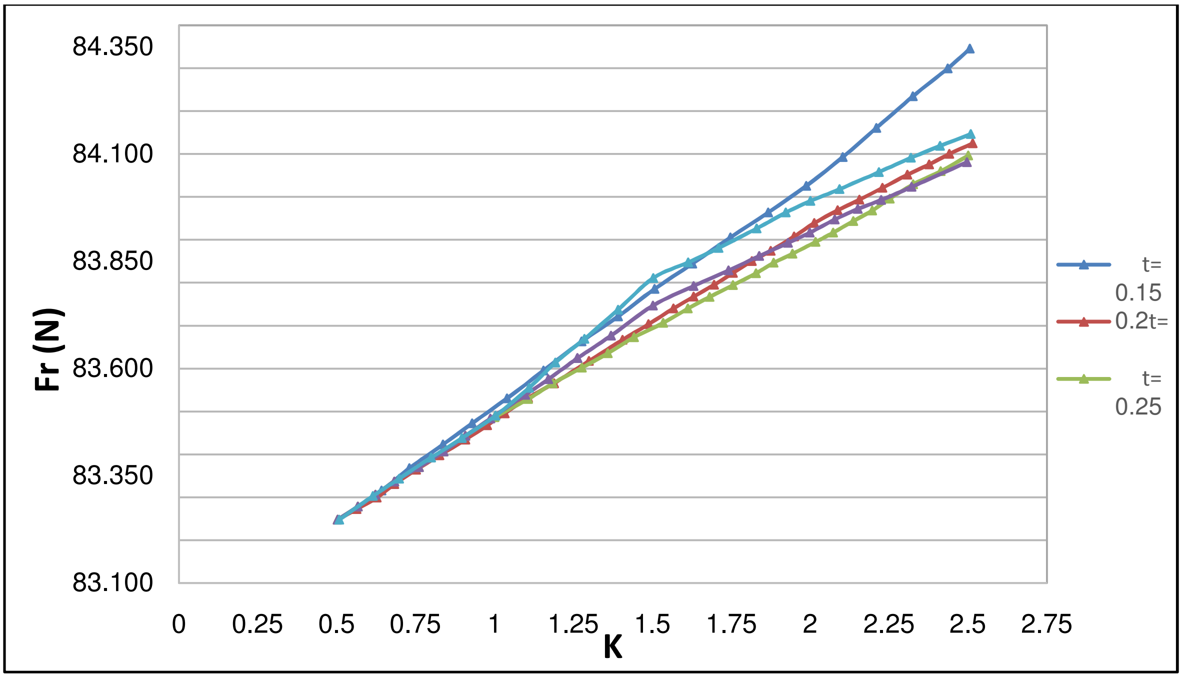

Figure 3: Friction Force versus Non-Dimensional Thermal Conductivity for Centerline Oil Croove

To be more specific the Thermal conductivity ($\overline{K}$) is fixed on (2.0) in order to investigate friction force values for different coating thickness. Figure 4 indicates the relationship of friction force against non-dimensional thickness for $\overline{K}$ =2.0. where the minimum friction force located at t=.25.

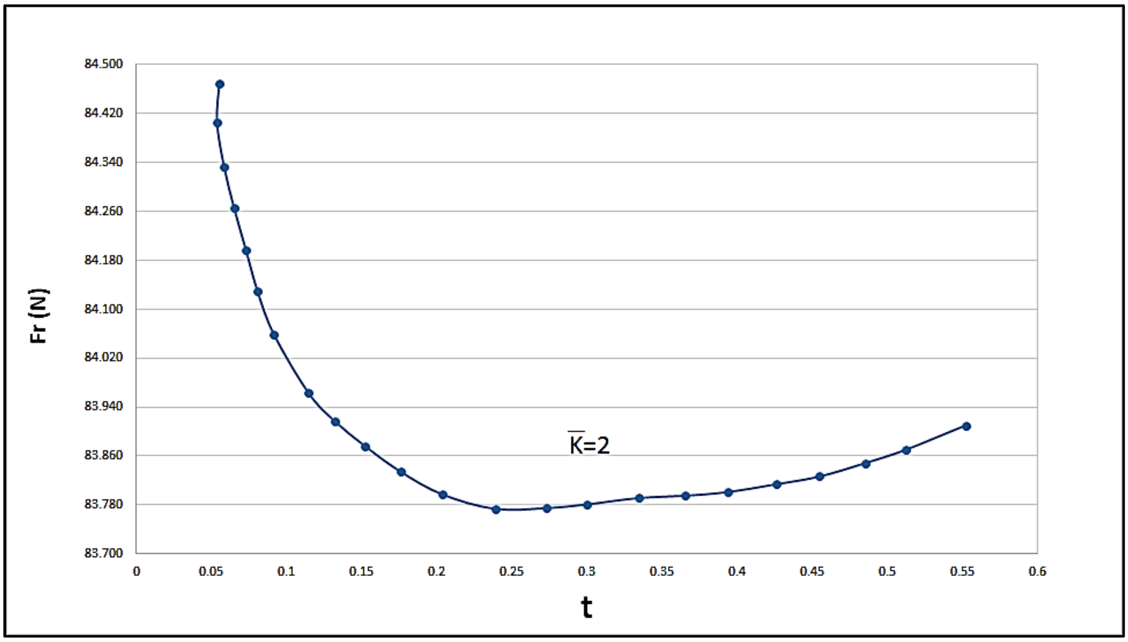

Figure 4: Friction force versus Non-Dimensional Coating Thickness for Centerline Oil Croove

Figure 5illustrate the relationship between non-dimensional maximum oil temperature and the eccentricity ratio (**ε**) with different non-dimensional thermal conductivity (K)= (2, 1 & 0.5) for coating thickness (t=0.25). Figure 5. shows

that the lower oil temperature located at maximum thermal conductivity (K=2)because the heat dissipates faster and the lubricant become cool. This feature demonstrates the reason of staying the oil cool in a condition of using white metal coating as due to its non-thermal conductivity relatively high comparison with a base metal of bush.

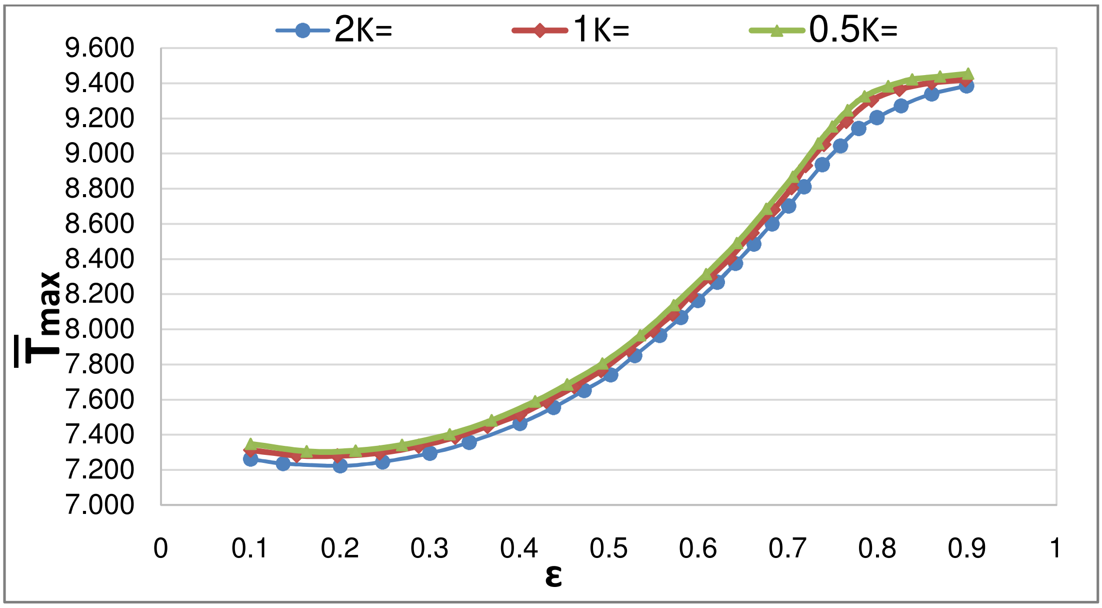

Figure 5: Non-Dimensional Maximuam Oil Temperature Versus Ecce. Ratio for center line Oil Croove, t=0.25

Figure 6presents the relation between the non–dimensional bush temperatures (Tb*) and eccentricity ratio (**ε**) with different non-dimensional thermal conductivity (K)= (2, 1 & 0.5) for coating thickness (t=0.25). The maximum non-dimensional thermal conductivity had been obtained from the lowest non-dimensional bush temperature for the same reason that mentioned in the discussion of Figure 5.

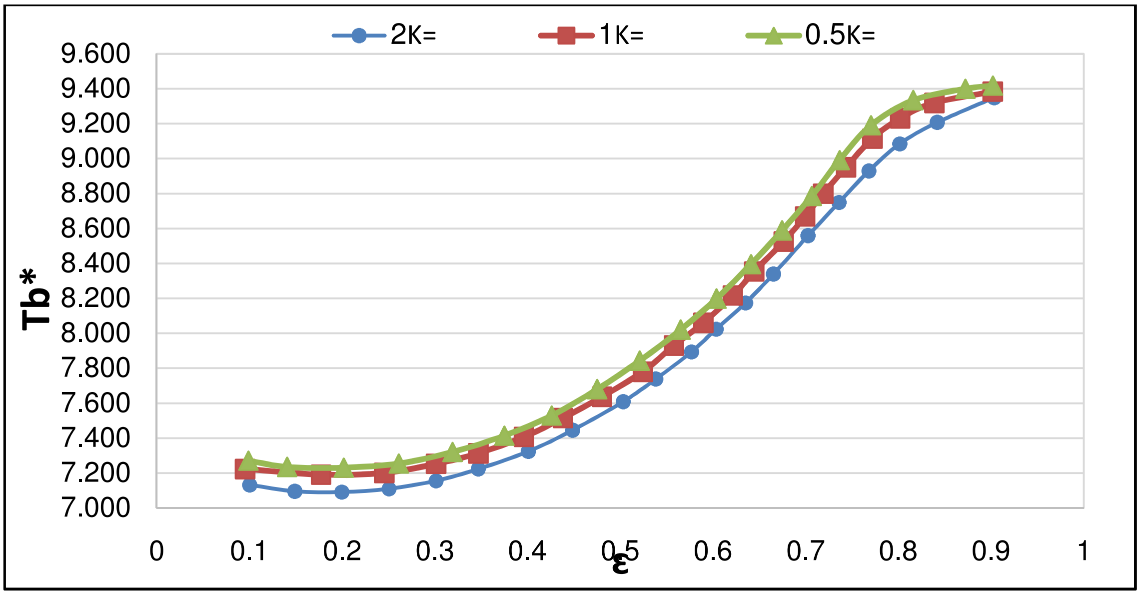

Figure 6: Non-Dimensional Maximuam Bush Temperature Versus Ecce. Ratio for Center line Oil Croove, t=0.25

Figure 7. demonstrates the relationship between the non-dimensional oil temperature $\overline{T}$across the oil film$\overline{H}$with different non-dimensional thermal conductivity (K)= (2, 1 & 0.5) for coating thickness (t=0.25). The cooling of oil is for maximum $thermal\ conductivity\ \overline{K}$. The effect of the white metal coating is more visible near the bush rather than near the journal surface due to the increasing of the heat dissipation through the white coating of the bush inner surface to the surrounding.

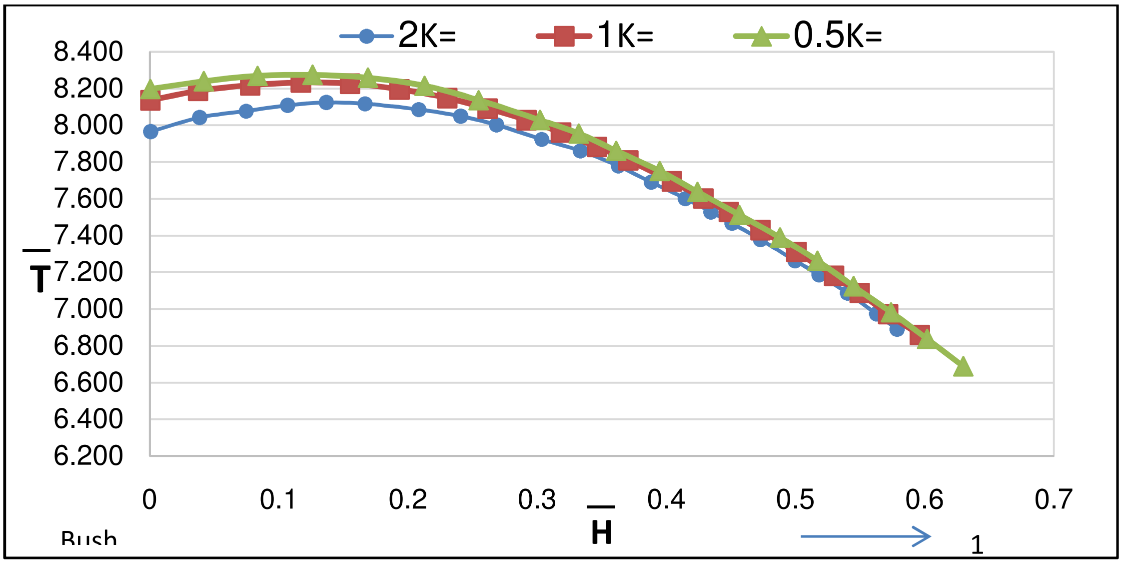

Figure 7: Non-Dimensional Maximum Oil Temperature in The Radial Direction for centerline Oil Croove, t=0.25

On the other hand, Figure 8 shows the relationship between the non–dimensional bush temperatures (Tb*) and Non-Dimensional Bush thickness for different non-dimensional thermal conductivity (K)= (2.5, 2, 1 & 0.5) for coating thickness (t=0.25). So, the effect of white metal coating at the inner bush surface is clearly reported as an increasing the heat dissipation from the oil across the 0.25 non-dimensional white metal thickness which increases the bush metal temperature.

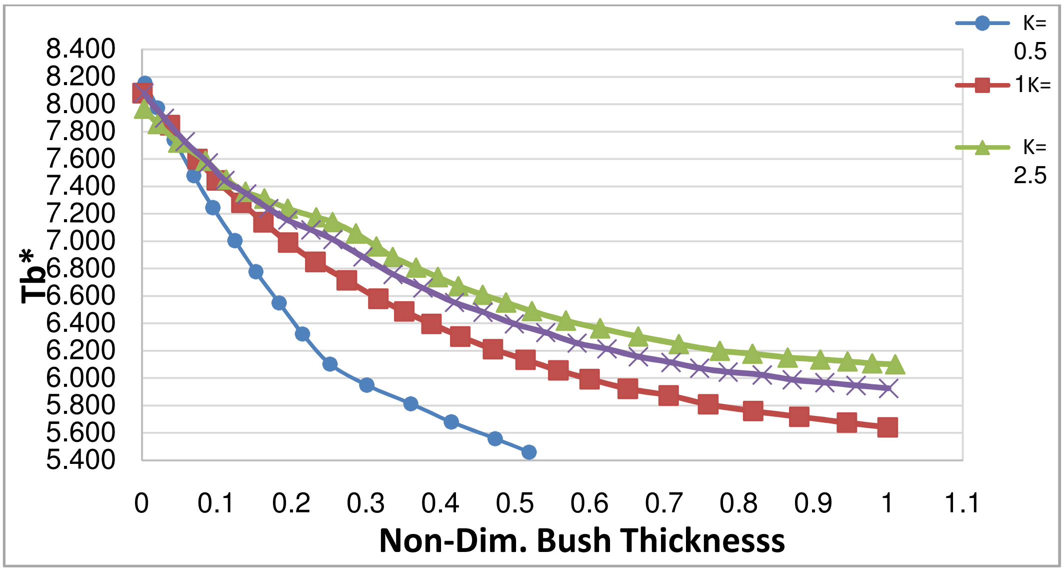

Figure 8: Non-Dimensional Bush Temperature in The Radial Direction for centerline Oil Croove, t=0.25

The load carrying capacity increase due to increasing the non-dimensional thermal conductivity as a result of heat dissipation via the white metal coating which decreases the Sommerfeld number (i.e. increase the load capacity) and decreases the oil temperature. Figure (9) shows the effect of non-dimensional white metal coating which is keep the oil cool and increase the load carrying capacity. It is clear that for the value below 0.1 of non-dimensional thermal conductivity the effect of various white metal thickness is almost close this means that the bush consists of one material. The increase of white metal thickness will lead to increases Sommerfeld number and the optimum value of non–dimensional white metal thickness is 0.25 which gives minimum friction force.

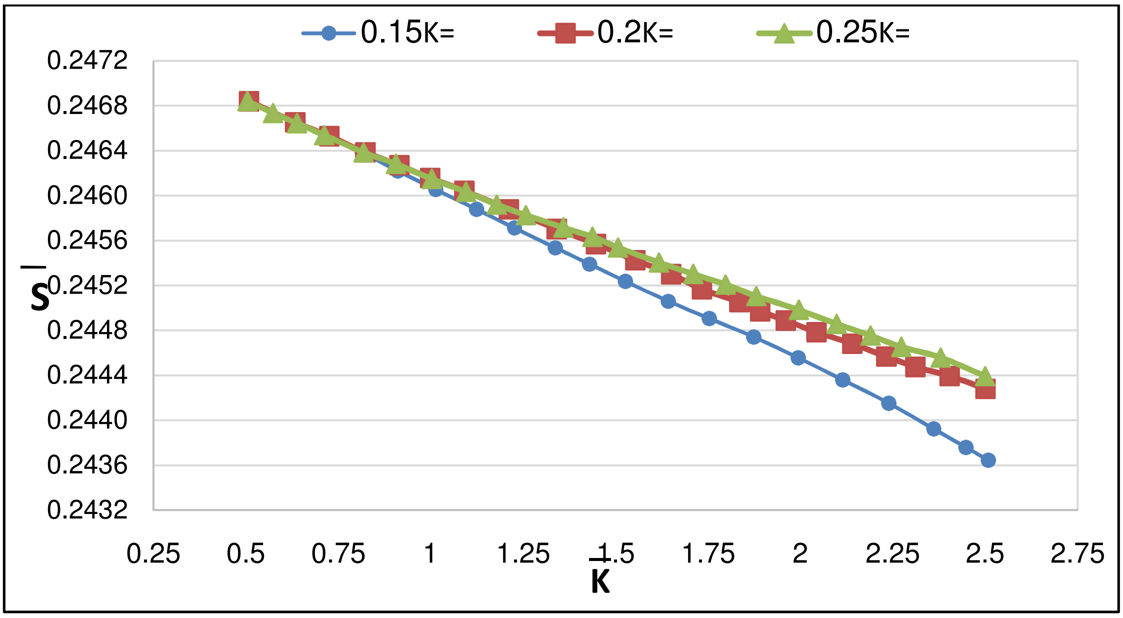

Figure 9: Sommerfeld Number Versus Non-Dimensional Thermal Conductivity for centerline Oil Croove

CONCLUSIONS

To conclude, based on the theoretical results, the rehabilitation of rotary parts by using white metal coating alloys has many effects as following:

Increasing of the non-dimensional thermal conductivity leads to increase the frictional force, non-dimensional maximum oiland bush temperatures, a non-dimensional coefficient of friction and decreasing Sommerfeld number. While the increasing of the non-dimensional coating thickness leads to increase in the Sommerfeld number and non-dimensional friction coefficient. The optimum non-dimensional coating thickness is 0.25 and the thickness effect is very clear near the bush inner surface. Through the white metal layer, the relation between non-dimensional thermal conductivity and the non-dimensional bush temperature is inverted.

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Terminologies

A,B,S	Viscosity Constants	-----
Bib	Biot number($h_b R/K_b$	-----
C	Radial clearance	m
	Specific heat of lubricant	J
	Specific heat of air	J
C	Equivalent specific heat of lubricant/air phase	J
D	Journal dia.	m
e	Eccentricity	m
F	Friction force	N
h	Film thickness	m
$\bar{H}$	Non-dim. Film thickness $\frac{h}{c}$	-------
$\bar{H}$	Non-dim. Minimum Film thickness $\frac{h}{c}$	-------

T	Lubricant temp.	
	Non- dimensional lubricant temp. $(T - T_{in)} C_p \rho \left(\frac{c}{R}\right)^2 / \mu_{in\ \omega}$	
	Eccentricity ratio	
	$Peclet\ number\ (\ K_O / C_p \rho \omega C^2$	
L	Bearing length	m
	Equivalent thermal conductivity of lubricant/air phase	W
$\bar{K}$	Non-dimensional thermal conductivity $^{K_c}/_{K_b}$	
K	Thermal conductivity of air	W
K	Thermal conductivity of bush base material	W
K	Thermal conductivity of coating material	W
K	Thermal conductivity of lubricant	W
	convection heat transfer coefficient	W